Exhibit 8

This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ ASHRAE Standard 51 but is included for information purposes.

APPENDIX A. Units of Measurement

A.1 System of Units. SI units (The International System of Units - Le Systéme International d'Unités)[24] are the primary units employed in this standard, with I-P units given as the secondary reference. SI units are based on the fundamental values of the International Bureau of Weights and Measures [25], and I-P values are based on the values of the National Institute of Standards and Technology which are, in turn, based on the values of the International Bureau. Appendix B provides conversion factors and coefficients for SI and other metric systems.

A.2 Basic Units. The unit of length is the meter *(m)* or millimeter *(mm)*; I-P units are the foot *(ft)* or inch *(in.)*. The unit of mass is the kilogram *(kg)*; the I-P unit is the pound-mass *(lbm)*. The unit of time is either the minute *(min)* or the second *(s)*. The unit of temperature is either the kelvin *(K)* or the degree Celsius *(°C)*; I-P units are the degree Rankine *(°R)* or the degree Fahrenheit *(°F)*. The unit of force is the newton *(N)*; the I-P unit is the pound *(lbf)*.

A.3 Airflow Rate and Velocity. The unit of airflow rate is the cubic meter per second *(m^3/s)*; the I-P unit is the cubic foot per minute *(cfm)*. The unit of velocity is the meter per second *(m/s)*; the I-P unit is the foot per minute *(fpm)*.

A.4 Pressure. The unit of pressure is either the Pascal *(Pa)* or the millimeter of mercury *(mm Hg)*; the I-P unit is either the inch water gauge *(in. wg)*, or the inch mercury column *(in. Hg)*. Values in *mm Hg* or in *in. Hg* shall be used only for barometric pressure measurements. The inch water gauge shall be based on a 1 inch column of distilled water at $68°F$ under standard gravity and a gas column balancing effect based on standard air. The millimeter of mercury shall be based on a 1 *mm* column of mercury at $0°C$ under standard gravity in vacuo. The inch of mercury shall be based on a 1 inch column of mercury at $32°F$ under standard gravity in vacuo.

A.5 Power, Energy, and Torque. The unit of power is the watt *(W)*; the I-P unit is the horsepower *(hp)*. The unit of energy is the joule *(J)*; the I-P unit is the foot pound *(ft• lbf)*. The unit of torque is the newton-meter *(N• m)*; the I-P unit is the pound inch *(lbf• in.)*.

A.6 Efficiency. Efficiencies are expressed on a per unit basis. Percentage values can be obtained by multiplying by 100.

A.7 Speed. There is no unit of rotational speed as such in the SI system of units. The commonly used unit in both systems is the revolution per minute *(rpm)*.

A.8 Gas Properties. The unit of density is the kilogram per cubic meter *(kg/m^3)*; the I-P unit is the pound-mass per cubic foot *(lbm/ft^3)*. The unit of viscosity is the Pascal second *(Pa• s)*; the I-P unit is the pound-mass per foot-second *(lbm/ft• s)*. The unit of gas constant is the joule per kilogram kelvin *(J/(kg• K))*; the I-P unit is the foot pound per pound mass degree Rankine *(ft• lbf/lbm• °R)*.

A.9 Dimensionless Groups. Various dimensionless quantities appear in the text. Any consistent system of units may be employed to evaluate these quantities unless a numerical factor is included, in which case units must be as specified.

This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ ASHRAE Standard 51 but is included for information purposes.

APPENDIX B. SI Conversions and Physical Constants

B.1 I-P Equivalents of SI Units [26].

Area

1 m^2	= 10.76 ft^2

Length

1 m (meter)	= 3.2808 ft

Mass

1 kg	= 2.2046 lbm

Temperature

1 K	= 1.8°R
t_c	= (t_F - 32)/1.8

Force

1 N	= 0.22481 lbf
1 kp (kilopond)	= 2.2046 lbf

Flow Rate

1 m^3/s	= 2118.9 cfm
1 m^3/h	= 0.58858 cfm

Velocity

1 m/s	= 196.85 fpm

Pressure

1 Pa (Pascal) at 20°C	= 0.0040264 $in.$ wg at 68°F
1 Pa (Pascal) at 0°C	= 0.00029530 $in.$ Hg at 32°F
1 Pa (Pascal) at 3.9°C	= 0.004015 $in.$ wg at 39°F

Power

1 W (watt)	= 0.0013410 hp

Energy

1 J (joule)	= 0.73756 $ft{\cdot}lbf$

Torque

1 $N{\cdot}m$	= 8.8507 $lbf{\cdot}in.$

Density

1 kg/m^3	= 0.062428 lbm/ft^3
1.200 kg/m^3 at 20°C	= 0.075 lbf/ft^3 at 68°F

Viscosity

1 $Pa{\cdot}s$	= 0.67197 $lbm/ft{\cdot}s$

Gas Constant

1 $J/(kg{\bullet}K)$	= 0.18586 $ft{\bullet}lbf/(lbm\cdot{}^{\circ}R)$

Gravitational Acceleration

9.80665 m/s^2	= 32.174 ft/s^2

B.2 SI Equivalents of I-P Units [26].

Area

1 ft^2	= 0.0929 m^2

Length

1 ft	= 0.30480 m

Mass

1 lbm =	0.45359 kg

Temperature

1°R	= K/1.8
t_F	= 1.8 t_c + 32

Force

1 lbf	= 4.4482 N

Flow Rate

1 cfm	= 0.00047195 m^3/s
1 cfm	= 1.6990 m^3/hr

Velocity

1 fpm	= 0.0050800 m/s

Pressure

1 $in.$ wg at 68°F	= 248.36 Pa at 20 °C
1 $in.$ wg at 39 °F	= 249.1 Pa at 3.9 °C
1 $in.$ Hg at 32 °F	= 3386.4 Pa at 0 °C

Power

1 hp (horsepower)	= 0.74570 kW

Energy

1 ft lbf	= 1.3558 J

Torque

1 lbf $in.$	= 0.11298 $N{\cdot}m$

Density

1 lbm/ft^3	= 16.018 kg/m^3
0.075 lbm/ft^3 at 68 °F	= 1.2 kg/m^3 at 20°C

Viscosity

1 $lbm/ft{\bullet}s$	= 1.4882 $Pa{\bullet}s$

Gas Constant

1 $ft{\bullet}lbf/(lbm\cdot{}^{\circ}R)$	= 5.3803 $J/(kg{\bullet}K)$

Gravitational Acceleration

32.174 ft/s^2	= 9.80665 m/s^2

B3. Physical Constants. The value of standard gravitational acceleration shall be taken as 9.80665 m/s^2 which corresponds to the value at mean sea level at 45° latitude; the I-P value is 32.1740 ft/s^2 [24]. The density of distilled water at saturation pressure shall be taken as 998.278 kg/m^3 at 20°C; the I-P value is 62.3205 lbm/ft^3 at 68°F [27]. The density of mercury at saturation pressure shall be taken as 13595.1 kg/m^3 at 0°C; the I-P value is 848.714 lbm/ft^3 at 32°F [27]. The specific weights in kg/m^3 (lbm/ft^3) of these fluids *in vacuo* under standard gravity are numerically equal to their densities at corresponding temperatures.

This Appendix is not part of ANSI/AMCA Standard 210 or ANSI/ ASHRAE Standard 51 but is included for information purposes.

APPENDIX C. Derivation of Equations

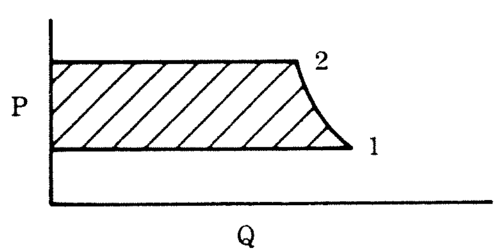
C.1 General. Various formulae appear in the standard. The origin of these formulae will be obvious to an engineer. Some, like the equations for α, β, P_t, P_s, and P_v, are algebraic expressions of fundamental definitions. Others, like the equations for p_e, μ, and C, are simply polynomials derived to fit the indicated data. Still others are derived from the equation of state, the Bernoulli equation, the equation of continuity, and other fundamental considerations. Only the less obvious formulae will be derived here.

C.2 Symbols. In the derivations which follow, certain symbols and notations are used in addition to those which are also used in the standard.

SYMBOL	DESCRIPTION	UNIT
H_i	Power Input to Impeller,	*W (hp)*
n	Polytropic Exponent	dimensionless
P	Absolute Total Pressure,	*Pa (in. wg)*

C.3 Fan Total Efficiency Equation. The values of the fan airflow rate, fan total pressure, and fan power input which are determined during a test are the compressible flow values for the fan speed and fan air density prevailing. A derivation of the fan total efficiency equation based on compressible flow values follows [23].

The process during compression may be plotted on a chart of absolute total pressure (P) versus airflow rate (Q). By using total pressure, all of the energy is accounted for including kinetic energy.



The fan power output (H_o) is proportional to the shaded area which leads to

$$H_o = \int_1^2 Q\,dP \qquad \text{Eq. C-1 SI}$$

$$H_o = \frac{1}{6362}\int_1^2 Q\,dP \qquad \text{Eq. C-1 I-P}$$

The compression process may be assumed to be polytropic for which, from thermodynamics,

$$Q = Q_1\left(\frac{P}{P_1}\right)^{-1/n} \qquad \text{Eq. C-2}$$

Substituting,

$$H_o = Q_1\int_1^2\left(\frac{P}{P_1}\right)^{-1/n} dP \qquad \text{Eq. C-3 SI}$$

$$H_o = \frac{Q_1}{6362}\int_1^2\left(\frac{P}{P_1}\right)^{-1/n} dP \qquad \text{Eq. C-3 I-P}$$

Integrating between limits,

$$H_o = Q_1 P_1\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \qquad \text{Eq. C-4 SI}$$

$$H_o = \frac{Q_1 P_1}{6362}\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \qquad \text{Eq. C-4 I-P}$$

But from the definition of fan total pressure (P_t),

$$P_1 = P_t/\left(\frac{P_2}{P_1} - 1\right) \qquad \text{Eq. C-5}$$

and the definition of fan total efficiency (η_t),

$$\eta_t = \frac{H_o}{H_i} \qquad \text{Eq. C-6}$$

it follows that

$$\eta_t = \frac{Q_1 P_t}{H_i}\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \Big/ \left(\frac{P_2}{P_1} - 1\right) \qquad \text{Eq. C-7 SI}$$

$$\eta_t = \frac{Q_1 P_t}{6362 H_i}\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \Big/ \left(\frac{P_2}{P_1} - 1\right) \qquad \text{Eq. C-7 I-P}$$

C.4 Compressibility Coefficient. The efficiency equation derived above can be rewritten

$$\eta_t = \frac{Q_1 P_t K_p}{H_i} \qquad \text{Eq. C-8 SI}$$

$$\eta_t = \frac{Q_1 P_t K_p}{6362\ H_i} \qquad \text{Eq. C-8 I-P}$$

where

$$K_p = \left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \Big/ \left(\frac{P_2}{P_1} - 1\right) \qquad \text{Eq. C-9}$$

This is one form of the compressibility coefficient.

C.5 Derivation of K_p in terms of x and z. The compressibility coefficient (K_p) was derived above in terms of the polytropic exponent (n) and the pressure ratio (P_2/P_1). The polytropic exponent can be evaluated from the isentropic exponent (γ) and the polytropic efficiency. The latter may be considered equal to the fan total efficiency for a fan without drive losses. From thermodynamics,

$$\left(\frac{n}{n-1}\right) = \eta_t\left(\frac{\gamma}{\gamma-1}\right) \qquad \text{Eq. C-10}$$

Two new coefficients (x and z), may be defined in terms of the information which is known from a fan test,

$$x = \frac{P_t}{P_1} \text{ and} \qquad \text{Eq. C-11}$$

$$z = \left(\frac{\gamma-1}{\gamma}\right)\left(\frac{H_i}{Q_1 P_1}\right) \qquad \text{Eq. C-12 SI}$$

$$z = \left(\frac{\gamma-1}{\gamma}\right)\left(\frac{6362 H_i}{Q_1 P_1}\right) \qquad \text{Eq. C-12 I-P}$$

Manipulating algebraically,

$$\left(\frac{\gamma}{\gamma-1}\right) = \frac{x}{z}\left(\frac{H_i}{Q_1 P_1}\right) \qquad \text{Eq. C-13 SI}$$

$$\left(\frac{\gamma}{\gamma-1}\right) = \frac{x}{z}\left(\frac{6362 H_i}{Q_1 P_1}\right) \text{ and} \qquad \text{Eq. C-13 I-P}$$

$$\frac{P_2}{P_1} = (1 + x) \qquad \text{Eq. C-14}$$

Substituting in the equation for K_p,

$$K_p = \frac{\eta_t \frac{x}{z}\left(\frac{H_i}{Q_1 P_t}\right)\left[(1+x)^{(\gamma-1)/\gamma\eta_t} - 1\right]}{(1+x) - 1} \qquad \text{Eq. C-15 SI}$$

$$K_p = \frac{\eta_t \frac{x}{z}\left(\frac{6362 H_i}{Q_1 P_t}\right)\left[(1+x)^{(\gamma-1)/\gamma\eta_t} - 1\right]}{(1+x) - 1} \qquad \text{Eq. C-15 I-P}$$

This reduces to

$$(1+z) = (1+x)^{(\gamma-1)/\gamma\eta_t} \qquad \text{Eq. C-16}$$

Taking logarithms and rearranging

$$\eta_t = \frac{\gamma-1}{\gamma}\,\frac{\ln(1+x)}{\ln(1+z)} \qquad \text{Eq. C-17}$$

Substituting

$$\eta_t = \left(\frac{Q_1 P_t}{H_i} \right) \frac{z}{x} \frac{\ln(1+x)}{\ln(1+x)} \qquad \text{Eq. C-18 SI}$$

$$\eta_t = \left(\frac{Q_1 P_t}{6362\, H_i} \right) \frac{z}{x} \frac{\ln(1+x)}{\ln(1+x)} \quad \text{and}$$

Eq. C-18 I-P

$$K_p = \left(\frac{z}{x} \right) \frac{\ln(1+x)}{\ln(1+z)} \qquad \text{Eq. C-19}$$

Since the coefficients x and z have been defined in terms of test quantities, direct solutions of K_p and η_t can be obtained for a test situation. An examination of x and z will reveal that x is the ratio of the total-pressure rise to the absolute total pressure at the inlet, and that z is the ratio of the total-temperature rise to the absolute total temperature at the inlet. If the total-temperature rise could be measured with sufficient accuracy, it could be used to determine z, but in most cases better accuracy is obtained from the other measurements.

C.6 Conversion Equations. The conversion equations which appear in Section 8.9.2 of the standard are simplified versions of the fan laws which are derived in Appendix D. Diameter ratio has been omitted in Section 8.9.2 because there is no need for size conversions in a test standard.

This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ ASHRAE Standard 51 but is included for information purposes.

APPENDIX D. Similarity and Fan Laws

D.1 Similarity. Two fans which are similar and have similar airflow conditions will have similar performance characteristics. The degree of similarity of the performance characteristics will depend on the degree of similarity of the fans and of the airflow through the fans.

D.1.1 Geometric Similarity. Complete geometric similarity requires that the ratios of all corresponding dimensions for the two fans be equal. This includes ratios of thicknesses, clearances, and roughnesses as well as all the other linear dimensions of the airflow passages. All corresponding angles must be equal.

D.1.2 Kinematic Similarity. Complete kinematic similarity requires that the ratios of all corresponding velocities for the two fans be equal. This includes the ratios of the magnitudes of corresponding velocities of the air and corresponding peripheral velocities of the impeller. The directions and points of application of all corresponding vectors must be the same.

D.1.3 Dynamic Similarity. Complete dynamic similarity requires that the ratios of all corresponding forces in the two fans be equal. This includes ratios of forces due to elasticity, dynamic viscosity, gravity, surface tension, and inertia as well as the pressure force. The directions and points of application of all corresponding vectors must be the same.

D.2 Symbols. In the derivations which follow, certain symbols and notations are used in addition to those which are used in the standard.

SYMBOL	DESCRIPTION	UNIT
n	Polytropic Exponent	dimensionless
P	Absolute Total Pressure,	*Pa (in. wg)*
$\overline{Q}$	Mean Airflow rate,	m^3/s *(cfm)*
(Prime)	Incompressible Value	--------------

D.3 Fan Laws for Incompressible Flow. The fan laws are the mathematical expressions of the similarity of performance for similar fans at similar flow conditions. These laws may be deduced from similarity considerations, dimensional analysis, or various other lines of reasoning (JORGENSEN, R., *Fan Engineering,* Buffalo Forge Company, Buffalo, New York, 1983, Chapter 12.)

D.3.1 Fan Total Efficiency. The efficiencies of completely similar fans at completely similar flow conditions are equal. This is the fundamental relationship of the fan laws. It emphasizes the fact that the fan laws can be applied only if the points of operation are similarly situated for the two fans being compared. The fan law equation for fan total efficiency (η_t) is, therefore,

$$\eta_{tc} = \eta_t \qquad \text{Eq. D-1}$$

D.3.2 Fan Airflow Rate. The requirements of kinematic similarity lead directly to the airflow rate relationships expressed by the fan laws. Air velocities must be proportional to peripheral velocities. Since airflow rate is proportional to air velocity times flow area, and since area is proportional to the square of any dimension, say impeller diameter (D), it follows that the fan law equation for fan airflow rate (Q) is

$$Q_c = Q\left(\frac{D_c}{D}\right)^3\left(\frac{N_c}{N}\right) \qquad \text{Eq. D-2}$$

D.3.3 Fan Total Pressure. The requirements of dynamic similarity lead directly to the pressure relationships expressed by the fan laws. Pressure forces must be proportional to inertia forces. Since inertia force per unit area is proportional to air density (ρ) and air velocity squared and since air velocity is proportional to peripheral speed, it follows that the fan law equation for fan total pressure (P_t) which is also force per unit area is

$$P_{tc} = P_t\left(\frac{D_c}{D}\right)^2\left(\frac{N_c}{N}\right)^2\left(\frac{\rho_c}{\rho}\right) \qquad \text{Eq. D-3}$$

D.3.4 Fan Power Input. For incompressible flow, the compressibility coefficient is unity and power input is proportional to airflow rate times pressure divided by efficiency. From the above fan law relationships for fan airflow rate, fan total pressure, and fan total efficiency, it follows that the fan law equation for fan power input (H) is

$$H_c = H\left(\frac{D_c}{D}\right)^5\left(\frac{N_c}{N}\right)^3\left(\frac{\rho_c}{\rho}\right) \qquad \text{Eq. D-4}$$

D.3.5 Fan Velocity Pressure. The fan law equation for fan velocity pressure (P_v) follows from that for fan total pressure,

$$P_{vc} = P_v\left(\frac{D_c}{D}\right)^2\left(\frac{N_c}{N}\right)^2\left(\frac{\rho_c}{\rho}\right) \qquad \text{Eq. D-5}$$

D.3.6 Fan Static Pressure. By definition,

$$P_{sc} = P_{tc} - P_{vc} \qquad \text{Eq. D-6}$$

D.3.7 Fan Static Efficiency. By definition,

$$\eta_{sc} = \eta_{tc} \left(\frac{P_{sc}}{P_{tc}} \right) \qquad \text{Eq. D-7}$$

D.4 Fan Laws for Compressible Flow. More general versions of the fan laws, which recognize the compressibility of air, can also be deduced from similarity considerations [23].

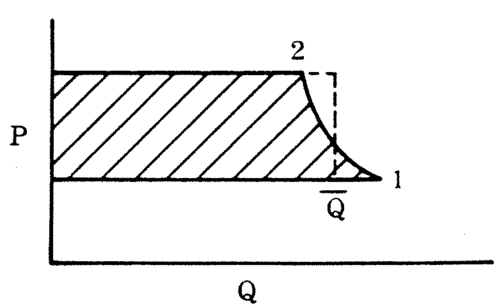
D.4.1 Fan Total Efficiency. It is obvious that airflow conditions can never be completely similar, even for two completely similar fans, if the degree of compression varies. Nevertheless, it is useful and convenient to assume that the fan law equation for fan total efficiency (η_t) need not be modified,

$$\eta_{tc} = \eta_t \qquad \text{Eq. D-8}$$

D.4.2 Fan Airflow Rate. Continuity requires that the mass airflow rate at the fan outlet equal that at the fan inlet. If the volumetric airflow rate at the inlet (Q_1) is proportional to peripheral speed, the volumetric airflow rate at the outlet (Q_2) cannot be proportional to peripheral speed or vice versa except for the same degree of compression. There is some average airflow rate which is proportional to peripheral speed and flow area. Since for a polytropic process, the airflow rate is an exponential function of pressure, the geometric mean of the airflow rates at the inlet and outlet will be a very close approximation of the average airflow rate $(\overline{Q})$. The geometric mean is the square root of the product of the two end values,

$$\overline{Q} \approx (Q_1 Q_2)^{1/2} \qquad \text{Eq. D-9}$$

the value ($\overline{Q}$) illustrated in the following diagram is the average airflow rate based on power output. This value yields the same power output as the polytropic process over the same range of pressures.



For the polytropic process,

$$H_o = \frac{Q_1 P_t K_p}{1} \qquad \text{Eq. D-10 SI}$$

$$H_o = \frac{Q_1 P_t K_p}{6362} \qquad \text{Eq. D-10 I-P}$$

For the rectangle,

$$H_o = \frac{\overline{Q} P_t}{1} \qquad \text{Eq. D-11 SI}$$

$$H_o = \frac{\overline{Q} P_t}{6362} \qquad \text{Eq. D-11 I-P}$$

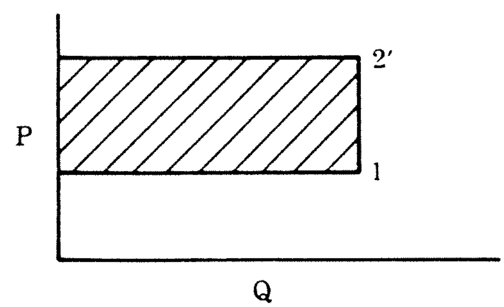
Therefore,

$$\overline{Q} = Q_1 K_p = Q K_p \qquad \text{Eq. D-12}$$

This average airflow rate can be substituted in D-2 to give the compressible flow fan law equation for fan airflow rate,

$$Q_c = Q \left(\frac{D_c}{D} \right)^3 \left(\frac{N_c}{N} \right) \left(\frac{K_p}{K_{pc}} \right) \qquad \text{Eq. D-13}$$

D.4.3 Fan Total Pressure. The incompressible flow fan laws are based on a process which can be diagramed as shown below.



The fan power output is proportional to the shaded area which leads to

$$H_o = Q_1 (P_2' - P_1) / 1 \qquad \text{Eq. D-14 SI}$$

$$H_o = Q_1 (P_2' - P_1) / 6362 \qquad \text{Eq. D-14 I-P}$$

Extending the definition of fan total pressure to the incompressible case

$$P_t' = (P_2' - P_1) \qquad \text{Eq. D-15}$$

Therefore,

$$H_o = Q_1 P_t' / 1 \qquad \text{Eq. D-16 SI}$$

$$H_o = Q_1 P_t' / 6362 \qquad \text{Eq. D-16 I-P}$$

For the same airflow rate (Q_1), absolute inlet pressure (P_1), and power output (H_o), the corresponding equation for compressible flow is

$$H_o = Q_1 P_t K_p / 1 \qquad \text{Eq. D-17 SI}$$

$$H_o = Q_1 P_t K_p / 6362 \qquad \text{Eq. D-17 I-P}$$

It follows that,

$$P_t' = P_t K_p \qquad \text{Eq. D-18}$$

The compressible flow fan law equation for fan total pressure can, therefore, be obtained by substitution,

$$P_{tc} = P_t \left(\frac{D_c}{D}\right)^2 \left(\frac{N_c}{N}\right)^2 \left(\frac{\rho_c}{\rho}\right) \left(\frac{K_p}{K_{pc}}\right) \qquad \text{Eq. D-19}$$

D.4.4 Fan Power Input. The equation for efficiency may be rearranged to give either

$$H = \frac{Q\, P_t K_p}{1\ \eta_t} \qquad \text{Eq.D-20 SI}$$

$$H = \frac{Q\, P_t K_p}{6362\ \eta_t} \text{ or} \qquad \text{Eq. D-20 I-P}$$

$$H_c = \frac{Q_c P_{tc} K_{pc}}{1\ \eta_{tc}} \qquad \text{Eq. D-21 SI}$$

$$H_c = \frac{Q_c P_{tc} K_{pc}}{6362\ \eta_{tc}} \qquad \text{Eq. D-21 I-P}$$

Combining and using the compressible flow fan law relationships for fan airflow rate, fan total pressure, and fan total efficiency, it follows that the compressible flow fan law equation for fan power input is,

$$H_c = H \left(\frac{D_c}{D}\right)^5 \left(\frac{N_c}{N}\right)^3 \left(\frac{\rho_c}{\rho}\right) \left(\frac{K_p}{K_{pc}}\right) \qquad \text{Eq. D-22}$$

D.4.5 Fan Velocity Pressure. By definition,

$$P_v = P_{v2} = \left(Q_2 / \sqrt{2}\ A_2\right)^2 \rho_2 \qquad \text{Eq. D-23 SI}$$

$$P_v = P_{v2} = \left(Q_2 / 1097\ A_2\right)^2 \rho_2 \qquad \text{Eq. D-23 I-P}$$

But from continuity,

$$\rho_2 Q_2 = \rho_1 Q_1 = \rho Q_1 \qquad \text{Eq. D-24}$$

Therefore,

$$P_v = \rho Q_1 Q_2 / (\sqrt{2}\ A_2)^2 \qquad \text{Eq. D-25 SI}$$

$$P_v = \rho Q_1 Q_2 / (1097\ A_2)^2 \qquad \text{Eq. D-25 I-P}$$

But from D-9 and D-12,

$$\overline{Q^2} = Q^2 K_p^2 \approx Q_1 Q_2 \qquad \text{Eq. D-26}$$

It follows that

$$P_v = \rho Q^2 K_p^2 / \left(\sqrt{2}\ A_2\right)^2 \qquad \text{Eq. D-27 SI}$$

$$P_v = \rho Q^2 K_p^2 / \left(1097\, A_2\right)^2 \qquad \text{Eq. D-27 I-P}$$

By similar reasoning,

$$P_{vc} = \rho_c Q_c^{\,2} K_{pc}^{\,2} \Big/ \left(\sqrt{2}\ A_{2c}\right)^2 \qquad \text{Eq. D-28 SI}$$

$$P_{vc} = \rho_c Q_c^{\,2} K_{pc}^{\,2} \Big/ \left(1097\ A_{2c}\right)^2 \qquad \text{Eq. D-28 I-P}$$

By using the compressible flow fan law relationships for fan airflow rate and the proportionality of outlet area to diameter squared, it follows that the compressible flow fan law equation for fan velocity pressure is

$$P_{vc} = P_v \left(\frac{D_c}{D}\right)^2 \left(\frac{N_c}{N}\right)^2 \left(\frac{\rho_c}{\rho}\right) \qquad \text{Eq. D-29}$$

D.4.6 Fan Static Pressure. By definition,

$$P_{sc} = P_{tc} - P_{vc} \qquad \text{Eq. D-30}$$

D.4.7 Fan Static Efficiency. By definition,

$$\eta_{sc} = \eta_{tc} \left(\frac{P_{sc}}{P_{tc}}\right) \qquad \text{Eq. D-31}$$

D.5 Fan Law deviations. Among the requirements for complete similarity are those for equal force ratios that lead to Reynolds and Mach number considerations.

D.5.1 Reynolds Number. There is some evidence that efficiency improves with an increase in Reynolds number (NIXON, R. A., *Examination of the Problem of Pump Scale Laws*, National Engineering Laboratory, Glasgow, Scotland, U.K., Paper 2D-1, 1967. (AMCA #1161)). However, that evidence is not considered to be sufficiently well documented to incorporate any rules in this Appendix. There is also some evidence that performance drops off with a significant decrease in Reynolds number. The fan laws should not be employed if it is suspected that the airflow regimes are significantly different because of a difference in Reynolds number.

D.5.2 Mach Number. There is evidence that choking occurs when the Mach number at any point in the flow passages approaches unity. Obviously, the fan laws should not be employed if this condition is suspected.

D.5.3 Bearing and Drive Losses. While there may be other similarity laws covering bearings and other drive elements, the fan laws cannot be used to predict bearing or drive losses. The correct procedure is to subtract the losses for the first condition, make fan law projections of power input for the corrected first condition to the second condition, and then add the bearing and drive losses for the second condition.

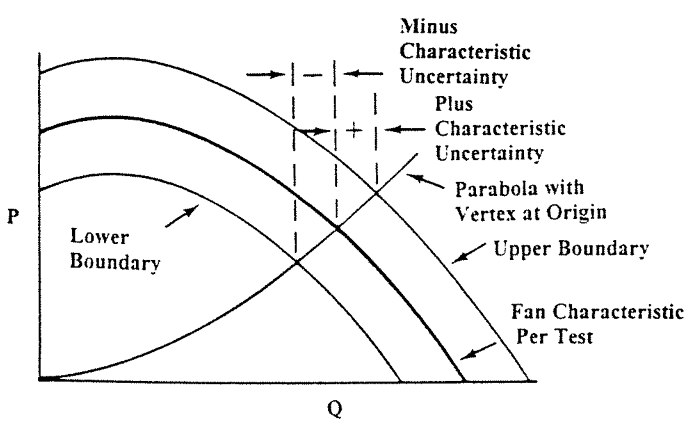
This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ ASHRAE Standard 51 but is included for information purposes.

APPENDIX E. Uncertainties Analysis [9]

E.1 General. This analysis is based on the assumption that fan performance can be treated as a statistical quantity and that the performances derived from repeated tests would have a normal distribution. The best estimate of the true performance would, therefore, be the mean results based on repeated observations at each point of operation. Since only one set of observations is specified in the standard, this analysis must deal with the uncertainties in the results obtained from a single set of observations.

The results of a fan test are a complex combination of variables which must be presented graphically according to the standard. In order to simplify this analysis, test results will be considered to be the curves of fan static pressure versus fan airflow rate and fan static efficiency versus fan airflow rate. Analysis of fan power input is unnecessary since it is a part of efficiency analysis. The findings from a total pressure analysis would be similar to those of a static pressure analysis.

The uncertainty in the results will be expressed in two parts, both of which will be based on the uncertainties in various measurements. That part dealing with the pressure versus airflow rate curve will be called the characteristic uncertainty and that dealing with the efficiency versus airflow rate curve will be called the efficiency uncertainty. The characteristic uncertainty can be defined with reference to the following diagram:



The diagram shows a plot of the fan static pressure versus fan airflow rate as determined by test per this standard. Surrounding this curve is a band of uncertainties, the boundaries of which are roughly parallel to the test curve. Also shown is a parabola with vertex at the origin that intersects the fan curve and both of the boundaries. The characteristic uncertainty is defined as the difference in airflow rate between the intersection of the parabola with the test curve and the intersections of the parabola with the boundaries. Typically, the absolute value of the characteristic uncertainty would be ± a certain number of m^3/s *(cfm)*. The relative characteristic uncertainty would be the absolute characteristic uncertainty divided by the airflow rate at the intersection with the test curve.

The absolute efficiency uncertainty is defined as the difference in efficiency between that at points corresponding to the above mentioned intersections with the boundaries and that at the above mentioned intersection with the fan test curve. Typically, this would be expressed as ± so many percent. The relative efficiency uncertainty would be the absolute efficiency uncertainty divided by the efficiency at the point corresponding to the above mentioned intersection with the test curve.

The accuracies specified in the standard are based on two standard deviations. This means that there should be a 95% probability that the uncertainty in any measurement will be less than the specified value. Since the characteristic uncertainty and the efficiency uncertainty are based on these measurements, there will be a 95% probability that these uncertainties will be less than the calculated value.

E.2 Symbols. In the analysis which follows, certain symbols and notations are used in addition to those which are used in the standard.

SYMBOL	QUANTITY
dP/dQ	Slope of Fan Characteristic
e_x	Per Unit Uncertainty in X
ΔX	Absolute Uncertainty in X
F_x	Correlation Factor for X

SUBSCRIPT	DESCRIPTION
A	Area
b	Barometric Pressure
C	Nozzle Discharge Coefficient
d	Dry-bulb Temperature
f	Pressure for Airflow Rate
g	Pressure for Fan Pressure
H	Fan Power Input
K	Character
m	Maximum

N	Fan Speed
o	Fan Power Output
P	Fan Pressure
Q	Fan Airflow Rate
T	Torque
V	Variable as Defined in Equation D-11
w	For Wet-bulb Depression, in t_w
X	Generalized Quantity (A, b, ...ρ)
η	Fan Efficiency
ρ	Fan Air Density

E.3 Measurement Uncertainties. The various measurement uncertainties which are permitted in the standard are listed below with some of the considerations that led to their adoption.

(1) Barometric pressure is easily measured within the ±170 *Pa* (±0.05 *in. Hg)* specified,

$$e_b = 1.70 / p_b \qquad \text{Eq. E-1 SI}$$

$$e_b = 0.05 / p_b \qquad \text{Eq. E-1 I-P}$$

(2) Dry-bulb temperature is easily measured within the ±1 °C (±2.0 °*F*) specified if there are no significant radiation sources,

$$e_d = 1.0 / \left(t_d + 273.15\right) \qquad \text{Eq. E-2 SI}$$

$$e_d = 2.0 / \left(t_d + 459.67\right) \qquad \text{Eq. E-2 I-P}$$

(3) Wet-bulb depression is easily measured within 3 °C (5.0 °*F*) if temperature measurements are within 1 °C (2.0 °*F*) and if air velocity is maintained in the specified range,

$$e_w = 3 / \left(t_d - t_w\right) \qquad \text{Eq. E-3 SI}$$

$$e_w = 5.0 / \left(t_d - t_w\right) \qquad \text{Eq. E-3 I-P}$$

(4) Fan speed requires careful measurement to hold the 0.5% tolerance specified,

$$e_N = 0.005 \qquad \text{Eq. E-4}$$

(5) Torque requires careful measurement to hold the 2.0% tolerance specified,

$$e_T = 0.02 \qquad \text{Eq. E-5}$$

(6) Nozzle discharge coefficients given in the standard have been obtained from ISO data and nozzles made to specifications should perform within a tolerance of 1.2% according to that data. A properly performed laboratory traverse is assumed to have equal accuracy,

$$e_c = 0.012 \qquad \text{Eq. E-6}$$

(7) The area at the flow measuring station will be within 0.5% when the diameter measurements are within the 0.2% specified,

$$e_A = 0.005 \qquad \text{Eq. E-7}$$

(8) The tolerance on the pressure measurement for determining flow rate is specified as 1% of the maximum reading during the test. This is easily obtained by using the specified calibration procedures. In addition, an allowance must be made for the mental averaging which is performed on fluctuating readings. This is estimated to be 1% of the reading. Using the subscript *m* to denote the condition for the maximum reading, a combined uncertainty can be written,

$$e_f = \left\{ (0.01)^2 + \left[0.01 \left(\frac{Q_m}{Q} \right)^2 \right]^2 \right\}^{1/2} \qquad \text{Eq. E-8}$$

(9) The pressure measurement for determining fan pressure is also subject to an instrument tolerance of 1% of the maximum reading and an averaging tolerance of 1% of the reading. In addition, there are various uncertainties which are related to the velocity pressure. A tolerance of 10% of the fan velocity pressure should cover the influence of yaw on pressure sensors, friction factor variances, and other possible effects,

$$e_g = \left\{ (0.01)^2 + \left[0.01 \left(\frac{P_m}{P} \right) \right]^2 + \left[0.1 \left(\frac{P_v}{P} \right) \right]^2 \right\}^{1/2} \qquad \text{Eq. E-9}$$

E.4 Combined Uncertainties. The uncertainties in the test performance are the result of using various values each of which is associated with an uncertainty. The combined uncertainty for each of the fan performance variables is given below. The characteristic uncertainty and the efficiency uncertainty are also given.

1) Fan air density involves the various psychrometric

measurements and the approximate formula

$$\rho = \frac{p_b V}{R\left(t_d + 273.15\right)} \qquad \text{Eq. E-10 SI}$$

$$\rho = \frac{70.73\, p_b V}{R\left(t_d + 459.67\right)} \qquad \text{Eq. E-10 I-P}$$

where

$$V = \left\{1.0 - 0.378\left[\frac{p_e}{p_b} - \frac{\left(t_d - t_w\right)}{1500}\right]\right\} \qquad \text{Eq. E-11 SI}$$

$$V = \left\{1.0 - 0.378\left[\frac{p_e}{p_b} - \frac{\left(t_d - t_w\right)}{2700}\right]\right\}$$

Eq. E-11 I-P

For random and independent uncertainties in products, the combined uncertainty is determined as follows:

$$\frac{\Delta\rho}{\rho} = \left\{\left(\frac{\Delta 1.0}{1}\right)^2 + \left(\frac{\Delta p_b}{p_b}\right)^2 + \left(\frac{\Delta V}{V}\right)^2 + \left(\frac{\Delta R}{R}\right)^2 + \left(\frac{\Delta t_d}{T_d + 273.15}\right)^2\right\}^{1/2}$$

Eq. E-12 SI

$$\frac{\Delta\rho}{\rho} = \left\{\left(\frac{\Delta 70.73}{70.73}\right)^2 + \left(\frac{\Delta p_b}{p_b}\right)^2 + \left(\frac{\Delta V}{V}\right)^2 + \left(\frac{\Delta R}{R}\right)^2 + \left(\frac{\Delta t_d}{T_d + 459.67}\right)^2\right\}^{1/2}$$

Eq. E-12 I-P

Assuming $\Delta 1.0$ ($\Delta 70.73$) and ΔR are both zero,

$$e_\rho = (e_b^2 + e_V^2 + e_d^2)^{1/2} \qquad \text{Eq. E-13}$$

It can be shown that

$$e_V^2 = [(0.00002349\, t_W - 0.0003204)\,\Delta(t_d - t_w)]^2$$

Eq. E-14 SI

$$e_V^2 = [(0.00000725\, t_W - 0.0000542)\,\Delta(t_d - t_w)]^2$$

Eq. E-14 I-P

(2) Fan airflow rate directly involves the area at the flow measuring station, the nozzle discharge coefficient, the square root of the pressure measurement for flow, and the square root of the air density. When making fan law conversions, fan speed has a first power effect on airflow rate. The effects of uncertainties in each of these variables can be expressed mathematically as follows, where e_{QX} is the uncertainty in airflow rate due to the uncertainty in X.

$$\left.\begin{array}{ll} e_{QA} = e_A & e_{QN} = e_N \\ e_{QC} = e_C & e_{Q\rho} = \dfrac{e_\rho}{2} \\ e_{Qf} = \dfrac{e_f}{2} & e_{QT} = 0 \\ e_{Qg} = 0 & \end{array}\right\} \qquad \text{Eq. E-15}$$

The uncertainty in the airflow rate only can be determined from the above uncertainties by combining,

$$e_Q = \left[e_c^2 + e_A^2 + \left(\frac{e_f}{2}\right)^2 + \left(\frac{e_\rho}{2}\right)^2 + e_N^2\right]^{1/2}$$

Eq. E-15A

(3) Fan pressure directly involves the pressure measurement for fan pressure. In addition, when making fan law conversions, air density has a first power effect on fan pressure while fan speed produces a second power effect. Mathematically,

$$\left.\begin{array}{ll} e_{PA} = 0 & e_{PN} = 2\, e_N \\ e_{PC} = 0 & e_{P\rho} = e_\rho \\ e_{Pf} = 0 & e_{PT} = 0 \\ e_{Pg} = e_g & \end{array}\right\} \qquad \text{Eq. E-16}$$

The uncertainty in the fan pressure only can be determined from the above uncertainties by combining,

$$e_P = \left[e_g^2 + e_\rho^2 + \left(2e_N\right)^2\right]^{1/2} \qquad \text{Eq. E-16A}$$

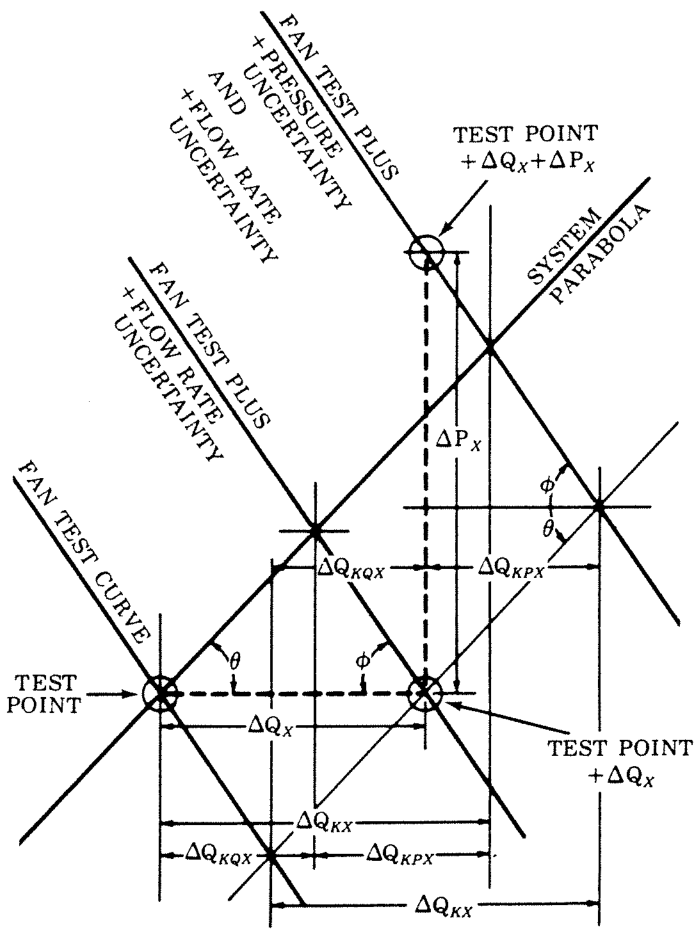
(4) Fan power input directly involves the torque and speed measurements. In addition, when making fan law conversions, density has a first power effect and speed a third power effect on fan power input. The net effect with respect to speed is second power. Mathematically,

$$\left.\begin{array}{ll} e_{HA} = 0 & e_{HN} = 2\, e_N \\ e_{HC} = 0 & e_{H\rho} = e_\rho \\ e_{Hf} = 0 & e_{HT} = e_T \\ e_{Hg} = 0 & \end{array}\right\} \qquad \text{Eq. E-17}$$

The uncertainty in the fan power input only can be determined from the above uncertainties by combining,

$$e_H = \left[e_T^{\,2} + e_\rho^{\,2} + \left(2e_N\right)^2\right]^{1/2} \qquad \text{Eq. E-17A}$$

(5) The uncertainties in the measurements for fan airflow rate and fan pressure create the characteristic uncertainty as defined in E.1. Assuming the uncertainties are small, the characteristic curves and parabola can be replaced by their tangents, and the effects of uncertainty in each measurement, (X), on the characteristic uncertainty can be determined. At a point (Q,P), the uncertainty in measurement (X) results in an uncertainty in Q and P of ΔQ_X and ΔP_X. For ΔQ_X,



$$\Delta Q_{KQX} \tan\theta = \left(\Delta Q_X - \Delta Q_{KQX}\right)\tan\phi \qquad \text{Eq. E-18}$$

$$\Delta Q_{KQX} = \Delta Q_X \left[\frac{\tan\phi}{\tan\theta + \tan\phi}\right] \qquad \text{Eq. E-19}$$

For ΔP_X,

$$\Delta Q_{KPX}\left(\tan\theta + \tan\phi\right) = \Delta P_X \qquad \text{Eq. E-20}$$

$$\Delta Q_{KPX} = \Delta P_X \left[\frac{1}{\tan\theta + \tan\phi}\right] \qquad \text{Eq. E-21}$$

Summing and simplifying by relating the tangents to the slopes of the parabola and the fan characteristic curve.

$$\Delta Q_{KX} = \Delta Q_{KQX} + \Delta Q_{KPX} \qquad \text{Eq. E-22}$$

$$\tan\theta = 2\left(\frac{P}{Q}\right) \text{ and} \qquad \text{Eq. E-23}$$

$$\tan\phi = -\left(\frac{dP}{dQ}\right) \qquad \text{Eq. E-24}$$

$$\Delta Q_{KX} = \Delta Q_X \left[\frac{-\left(\frac{dP}{dQ}\right)}{2\left(\frac{P}{Q}\right) - \left(\frac{dP}{dQ}\right)}\right] + \Delta P_X \left[\frac{1}{2\left(\frac{P}{Q}\right) - \left(\frac{dP}{dQ}\right)}\right] \qquad \text{Eq. E-25}$$

$$e_{KX} = e_{QX} \left[\frac{-\left(\frac{dP}{dQ}\right)}{2\left(\frac{P}{Q}\right) - \left(\frac{dP}{dQ}\right)}\right] + \frac{e_{PPX}}{2} \left[\frac{2\left(\frac{P}{Q}\right)}{2\left(\frac{P}{Q}\right) - \left(\frac{dP}{dQ}\right)}\right] \qquad \text{Eq. E-26}$$

Introducing correlation factors

$$F_Q = \left[\frac{-\left(\frac{dP}{dQ}\right)}{2\left(\frac{P}{Q}\right) - \left(\frac{dP}{dQ}\right)} \right] \text{ and} \qquad \text{Eq. E-27}$$

$$F_P = \left[\frac{2\left(\frac{P}{Q}\right)}{2\left(\frac{P}{Q}\right) - \left(\frac{dP}{dQ}\right)} \right] \qquad \text{Eq. E-28}$$

$$e_{KX} = e_{QX} F_Q + \left(\frac{e_{PX}}{2}\right) F_P \qquad \text{Eq. E-29}$$

Combining equations E-15, E-16, and E-29,

$$e_{KA} = e_A F_Q,\ e_{Kg} = \left(\frac{e_g}{2}\right) F_p$$

$$e_{KC} = e_C F_Q,\ e_{KN} = e_N \left(F_Q + F_P\right) \qquad \text{Eq. E-30}$$

$$e_{Kf} = \left(\frac{e_f}{2}\right) F_Q,\ e_{K\rho} = \frac{e_\rho}{2} \left(F_Q + F_P\right)$$

Assuming these uncertainties are independent, they can be combined for the characteristic uncertainty as follows, noting that $F_Q + F_p = 1$,

$$e_K = \left\{ \left(\frac{e_\rho}{2}\right)^2 + e_N^{\ 2} + F_P^{\ 2}\left(\frac{e_g}{2}\right)^2 + F_Q^{\ 2}\left[e_C^{\ 2} + e_A^{\ 2} + \left(\frac{e_f}{2}\right)^2\right]\right\}^{1/2} \qquad \text{Eq. E-31}$$

(6) Fan power output is proportional to the third power of airflow rate along a system characteristic. Therefore,

$$e_o = 3\ e_K \qquad \text{Eq. E-32}$$

(7) Fan efficiency uncertainty was defined in E.1. Using the above noted correlation factors and recombining the components,

$$e_\eta = \left\{ \left(\frac{e_\rho}{2}\right)^2 + e_N^{\ 2} + e_T^{\ 2} + 9\left[F_p^{\ 2}\left(\frac{e_g}{2}\right)^2 + F_Q^{\ 2}\left(e_C^{\ 2} + e_A^{\ 2} + \left\{\frac{e_f}{2}\right\}^2\right)\right]\right\}^{1/2} \qquad \text{Eq. E-33}$$

E.5 Example. The characteristic test curve for a typical backward-curve centrifugal fan was normalized on the basis of shut-off pressure and free-delivery airflow rate. The resultant curve is shown in Figure E-1.

An uncertainty analysis based on this curve and the maximum allowable measurement tolerances follows:

(1) The maximum allowable measurement tolerances can be determined using the information from Section E.3. Where appropriate, lowest expected barometer and temperature for a laboratory at sea level are assumed.

Per unit uncertainties are:

$$e_b = [0.2/96.5] = 0.0021 \qquad \text{Eq. E-34 SI}$$
$$e_b = [0.05/28.5] = 0.0018 \qquad \text{Eq. E-34 I-P}$$

$$e_d = [1.0/(15.5 + 273.2)] = 0.0035 \qquad \text{Eq. E-35 SI}$$
$$e_d = [2.0/(60 + 459.7)] = 0.0038 \qquad \text{Eq. E-35 I-P}$$

$$e_w = [3.0/(15.5 - 10)] = 0.545 \qquad \text{Eq. E-36 SI}$$
$$e_w = [5.0/(60 - 50)] = 0.5 \qquad \text{Eq. E-36 I-P}$$

$$e_N = 0.005$$

$$e_T = 0.02$$

$$e_C = 0.012$$

$$e_A = 0.005$$

$$e_f = \left\{(0.01)^2 + \left[0.01\left(\frac{Q_m}{Q}\right)^2\right]^2\right\}^{1/2} \text{ and}$$

$$e_g = \left\{(0.01)^2 + \left[0.01\left(\frac{P_m}{P}\right)\right]^2 + \left[0.1\left(\frac{P_v}{P}\right)\right]^2\right\}^{1/2}$$

Eq. E-37

Note that e_f and e_g vary with point of operation. In this example, the values of Q_m, Q, P_m, and P are taken from Figure E-1. The velocity pressure at free delivery is taken to be 20% of the maximum static pressure.

(2) The various combined uncertainties and factors can be determined using the information from Section E.4. To illustrate, the per unit uncertainty in air density will be calculated:

$$e_\rho = (e_b^2 + e_V^2 + e_d^2)^{1/2}$$

$e_b^2 = [0.2/96.5]^2 = 0.0000043$ Eq. E-38 SI

$e_b^2 = [0.05/28.5]^2 = 0.00000308$ Eq. E-38 I-P

$e_V^2 = [(0.00002349 \times 10 + 0.0003204)\ (3.0)]^2 = 0.0000028$ Eq. E-39 SI

$e_V^2 = [(0.00000725 \times 50 + 0.0000542)\ (5.0)]^2 = 0.00000238$ Eq. E-39 I-P

$e_d^2 = [1.0/(15.5 + 273.2)]^2 = 0.000012$ Eq. E-40 SI

$e_d^2 = [2.0/(60 + 459.7)]^2 = 0.0000148$ Eq. E-40 I-P

and

$e_\rho = 0.0045$

This is the expected accuracy for a laboratory at sea level. For extremes of altitude and wet-bulb temperatures, the limit is $e_\rho = 0.005$.

(3) The characteristic uncertainty and the efficiency uncertainty can be calculated for various points of operation as indicated in Table E-1.

The values of Q, P, and $-(dP/dQ)$ have been read directly from the normalized fan curve. The results have been plotted as curves of per unit uncertainty versus airflow rate in Figure E-2.

E.6 Summary. The example is based on uncertainties which, in turn, are based on 95% confidence limits. Accordingly, the results of 95% of all tests will be better than indicated. Per unit uncertainties of one half those indicated will be achieved in 68% of all tests while indicated per unit uncertainties will be exceeded in 5% of all tests. The following conclusions can be drawn from the above example.

(1) The characteristic uncertainty for the specified tolerances is about 1% near the best efficiency point and approaches 2% at free delivery. The uncertainty also increases rapidly as shutoff is approached.

(2) The fan efficiency uncertainty is about 3% near the best efficiency point and exceeds 5% at free delivery. The uncertainty increases rapidly near shutoff.

(3) Psychrometric measurement uncertainties have very little effect on overall accuracy. Calibration corrections are unnecessary in most cases.

(4) The nozzle discharge coefficient uncertainty has a very significant effect on overall accuracy. The 1.2% tolerance specified was based on the current state of the art. Any significant improvement in the accuracy of test results will depend on further work to reduce the uncertainty of this quantity.

(5) While the example was based on a typical characteristic for a backward-curve centrifugal fan, analyses of different characteristics for other fan types will yield sufficiently similar results that the same conclusion can be drawn.

(6) This analysis has been limited to a study of measurement uncertainties in laboratory setups. Other factors may have an equal or greater effect on fan performance. The results of an on-site test may deviate from predicted values because of additional uncertainties in measurements such as poor approach conditions to measuring stations. Deviations may also be due to conditions affecting the airflow into or out of the fan which, in turn, affects the ability of the fan to perform. Differences in construction, which arise from manufacturing tolerances, may cause full-scale test performance to deviate from model performance.

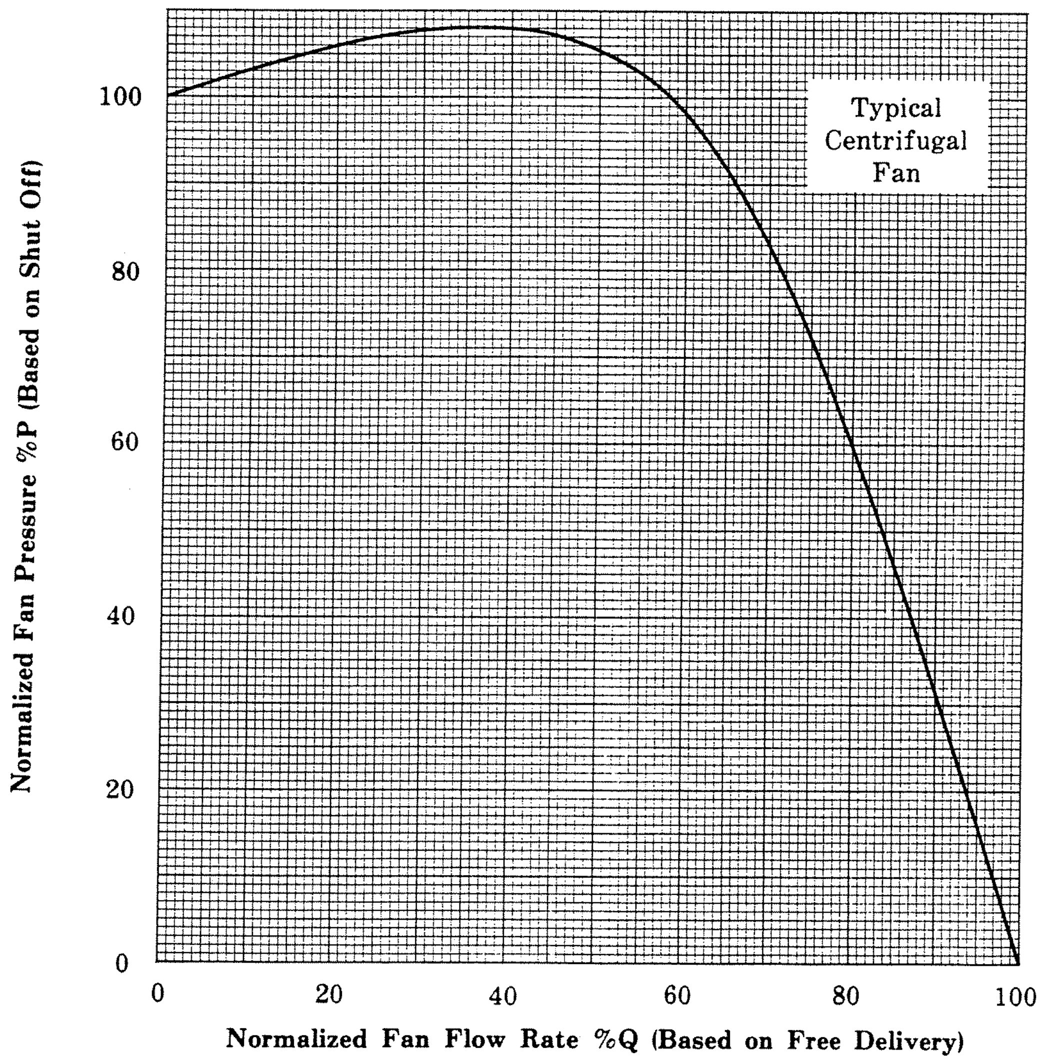


Figure E-1 Normalized Fan Performance Curve

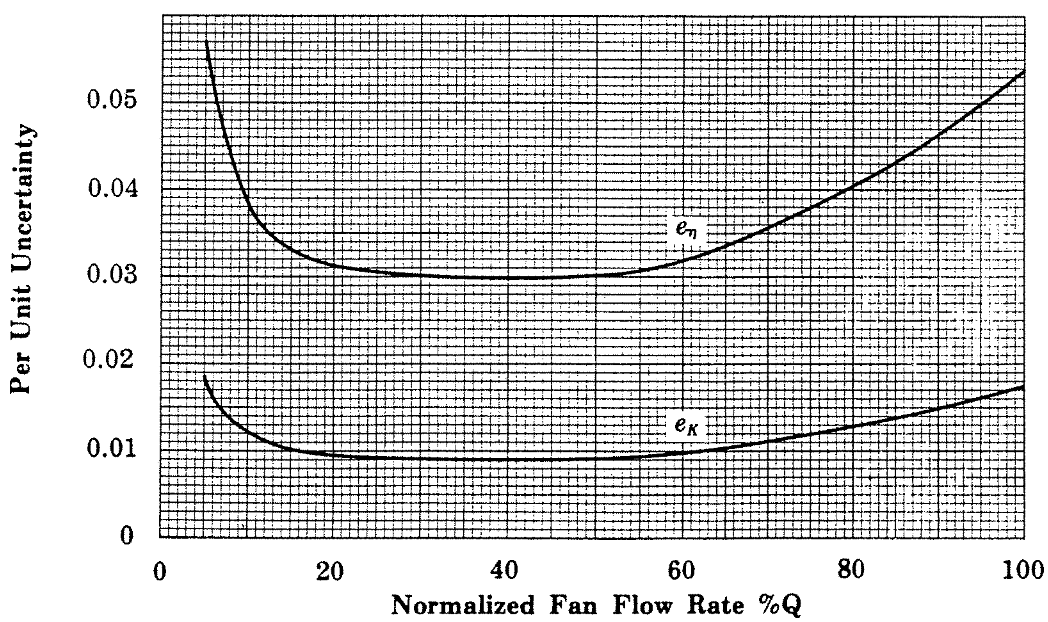


Figure E-2 Normalized Test Results Uncertainties

TABLE E-1 TABULATION FOR UNCERTAINTY ANALYSIS OF FIGURE E-1

$\%Q$	$\%P$	$\left(\frac{dP}{dQ}\right)$	F_P	F_Q	$\left[\left(\frac{e_\rho}{2}\right)^2 + e_N^2\right]$	$\left[F_P^2\left(\frac{e_g}{2}\right)^2\right]$	$\left[F_Q^2\left(e_c^2 + e_A^2 + \frac{e_f^2}{4}\right)\right]$	e_K	e_η
99	3.2	3.215	.01971	.98029	31.2 E-06	53.5 E-06	211.4 E-06	.0172	.0531
95	16.0	3.075	.09873	.90127	31.2 E-06	47.5 E-06	182.5 E-06	.0162	.0500
90	31.5	2.900	.19444	.80556	31.2 E-06	41.2 E-06	150.6 E-06	.0149	.0464
85	46.0	2.700	.28616	.71384	31.2 E-06	36.8 E-06	123.2 E-06	.0138	.0433
80	59.5	2.500	.37304	.62696	31.2 E-06	33.7 E-06	100.2 E-06	.0129	.0405
75	72.0	2.275	.45769	.54231	31.2 E-06	31.9 E-06	80.2 E-06	.0120	.0379
70	82.7	1.950	.54786	.45214	31.2 E-06	32.5 E-06	60.9 E-06	.0112	.0357
65	91.2	1.575	.64051	.35949	31.2 E-06	34.9 E-06	43.1 E-06	.0105	.0337
60	98.0	1.150	.73962	.26038	31.2 E-06	38.8 E-06	26.2 E-06	.0098	.0319
55	102.6	.800	.82343	.17657	31.2 E-06	42.6 E-06	14.5 E-06	.0094	.0307
50	105.3	.500	.89389	.10611	31.2 E-06	46.2 E-06	6.6 E-06	.0092	.0301
45	107.0	.250	.95006	.04994	31.2 E-06	49.3 E-06	2.0 E-06	.0091	.0299
40	107.9	.050	.99082	.00918	31.2 E-06	51.6 E-06	0 E-06	.0091	.0299
35	108.0	- .025	1.00407	- .00407	31.2 E-06	51.9 E-06	0 E-06	.0091	.0300
30	107.6	- .100	1.01414	- .01414	31.2 E-06	52.4 E-06	0.6 E-06	.0092	.0301
25	107.0	- .175	1.02087	- .02087	31.2 E-06	53.0 E-06	2.8 E-06	.0093	.0306
20	106.0	- .225	1.02169	- .02169	31.2 E-06	53.5 E-06	7.4 E-06	.0096	.0313
15	104.7	- .275	1.02009	- .02009	31.2 E-06	53.7 E-06	20.0 E-06	.0102	.0331
10	103.2	- .325	1.01600	- .01600	31.2 E-06	54.0 E-06	64.0 E-06	.0122	.0386
5	101.6	- .325	1.00806	- .00806	31.2 E-06	54.1 E-06	259.8 E-06	.0186	.0571

APPENDIX F

This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ASHRAE Standard 51 but is included for information purposes.

APPENDIX F. ITERATIVE PROCEDURE

To obtain the value of C to be used in calculating the chamber nozzle airflow rate in 9.3.2.7, an iteration process is used. A calculated value of Re is made and using an estimated value of C. The calculated value of Re is then used to recalculate C until the difference between two successive trial values of C is < 0.001, at which point the last trial value of C is taken as the value to be used in calculating chamber nozzle volume. In the following example, the first estimate of Re is made using an estimated value of Ce = 0.99. It is suggested that calculations be carried out to at least 5 decimal places.

EXAMPLE ITERATION

F.1 Example Iteration (SI System of Units)

Iteration 1:

Step 1-1: Calculate Re, using

$$Re = \frac{\sqrt{2}}{\mu_6} Ce\ D_6\ Y \sqrt{\frac{\Delta P\ \rho_5}{1 - E\beta^4}} \qquad \text{Eq. F-1 SI}$$

$$? = \frac{1097}{60\ \mu_6} Ce\ D_6\ Y \sqrt{\left(\frac{\Delta P\ \rho_5}{[1 - E\ \beta^4}\right.} \qquad \text{Eq. F-1 I-P}$$

where:

μ_6 = 1.819 E-05 *Pa•s* (1.222E-05 *lbm/ft•s*)
Ce = 0.95 (estimated) (0.95 estimated)
D_6 = 0.15 *m* (6 *in.* = 0.5 *ft*)
Y = 0.998 (calculate per Section 9.3.2)
ΔP = 250 *Pa* (1.005 *in. wg*)
ρ_5 = 1.14 kg/m^3 (0.0711 lbm/ft^3)
(1-E β^4) = 1 for iteration purposes

Then:

$$Re_1 = \frac{\sqrt{2}}{\mu_6} Ce\ D_6\ Y \sqrt{\Delta P\ \rho_5}$$

$$= \frac{\sqrt{2}}{1.819\mathrm{E}-05} (0.95)\ (0.15)\ (0.998) \sqrt{(250)\ (1.14)}$$

$$= 186\ 660$$

$$Re_1 = \frac{1097}{60\ \mu_6} Ce\ D_6\ Y \sqrt{\Delta P\, \rho_5}$$

$$= \frac{1097}{(60)\, 1.222E-05} (0.95)(0.5)(0.998) \sqrt{(1.005)(0.0711)}$$

$$= 189\ 595$$

Step 1-2: Using Eq. F-2 (Eq. 8.19), calculate Ce_1, using Re_1 from the previous step, assuming that L/D = 0.6:

$$Ce_1 = 0.9986 - \frac{7.006}{\sqrt{Re}} + \frac{134.6}{Re}$$

$$= 0.9986 - \frac{7.006}{\sqrt{186660}} + \frac{134.6}{186660}$$

$$= 0.98311 \qquad \text{Eq. F-2 SI}$$

Check: C_l - Ce = 0.98311 - 0.95
= 0.03311
Since 0.0331 > 0.001, another iteration is required.

$$Ce_1 = 0.9986 - \frac{7.006}{\sqrt{Re}} + \frac{134.6}{Re}$$

$$= 0.9986 - \frac{7.006}{\sqrt{195595}} + \frac{134.6}{189595}$$

$$= 0.98322 \qquad \text{Eq. F-2 I-P}$$

Check: Ce_1 - Ce = 0.98322-0.95
= 0.03322

Since 0.03322 > 0.001, another iteration is required.

Iteration 2:

Step 2-1: Re-estimate Re, using Ce_1:

$$Re_2 = \frac{\sqrt{2}}{\mu_6} Ce_1\ D_6\ Y \sqrt{\Delta P\, \rho_5}$$

$$= \frac{\sqrt{2}}{1.819E-05} (0.98311)\ (0.15)\ (0.998) \sqrt{(250)\ (1.14)}$$

$$= 193\ 165 \qquad \text{Eq. F-3 SI}$$

$Re_2 = 199{,}397\ Ce_1$

$= 199{,}397\ 0.984$

$$Re_2 = \frac{1097}{60\ \mu_6}\ Ce_1\ D_6\ Y\ \sqrt{\Delta P\ \rho_5}$$

$$= \frac{1097}{(60)\,1.22E{-05}}(0.98322)(0.5)(0.998)$$

$$\sqrt{(1.005)(0.0711)}$$

$= 196\ 225$ Eq. F-3 I-P

Step 2-2: Recalculate C, using Re_2:

$$Ce_2 = 0.9986 - \frac{7.006}{\sqrt{Re_2}} + \frac{134.6}{Re_2}$$

$$= 0.9986 - \frac{7.006}{\sqrt{193\,165}} + \frac{134.6}{193\,165}$$

$= 0.98336$ Eq. F-4 SI

Check: $C_2 - C_1 = 0.98336 - 0.98311$

$= 0.00025$

Since $0.00025 < .001$, $C = C_2 = 0.98336$

$$Ce_2 = 0.9986 - \frac{7.006}{\sqrt{Re_2}} + \frac{134.6}{Re_2}$$

$$= 0.9986 - \frac{7.006}{\sqrt{196\ 225}} + \frac{134.6}{196\ 225}$$

$= 0.98347$ Eq. F-4 I-P

Check: $Ce_2 - C_I = 0.98344 - 0.98322$

$= 0.00025$

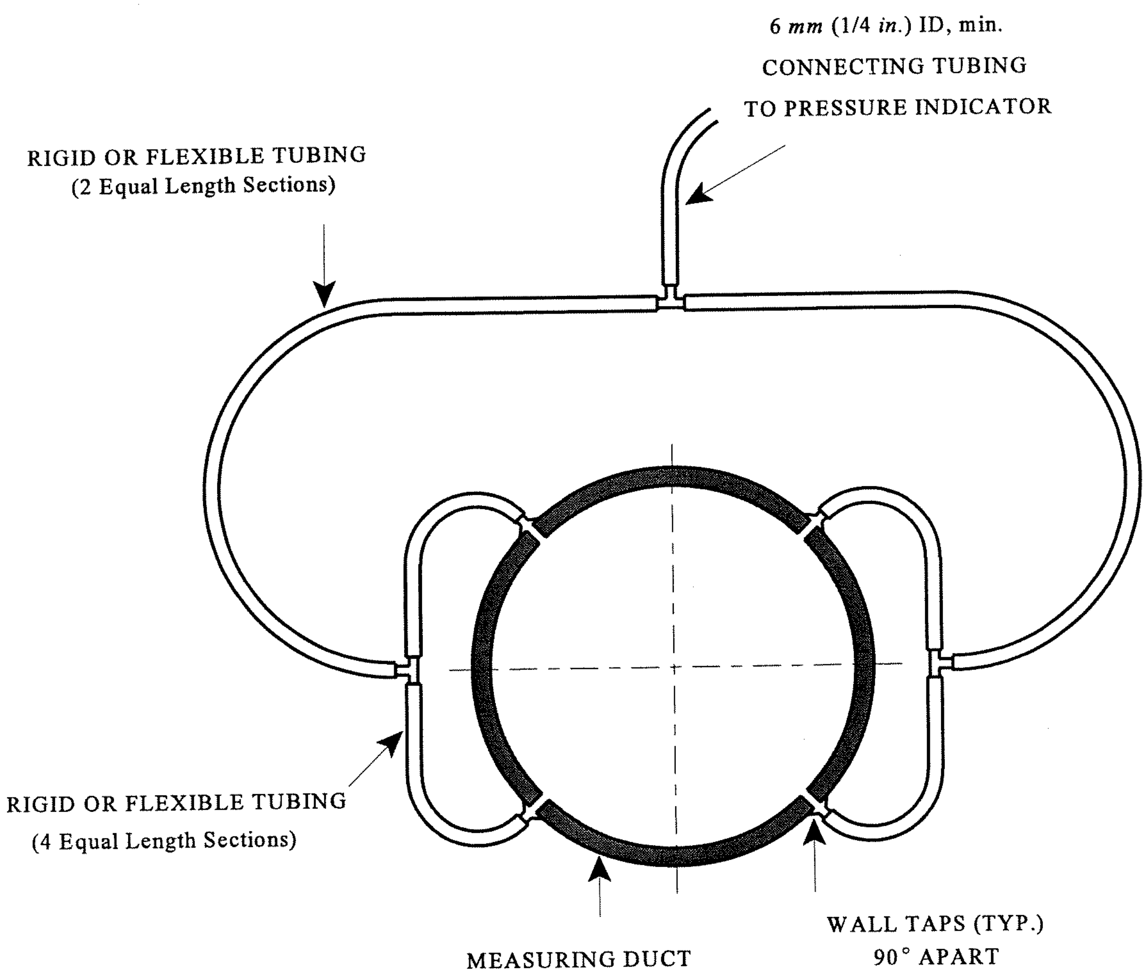
Since $0.00025 < 0.001$, $C = C_2 = 0.98347$

This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ASHRAE Standard 51 but is included for information purposes.

APPENDIX G. TUBING

Large tubing should be used to help prevent blockage from dust, water, ice, etc. Accumulations of dirt are especially noticeable in the bottom of round ducts; it is recommended that duct piezometer fittings be located at 45° from the horizontal. Tubing longer than 1.5 *m* (5 *ft*) should be a minimum of 6 *mm* (1/4 *in.*) inside diameter to avoid long pressure response times. When pressure response times are long, inspect for possible blockage. Hollow flexible tubing used to connect measurement devices to measurement locations should be of relatively large inside diameter. The larger size is helpful in preventing blockage due to dust, water, ice, etc.

Piezometer connections to a round duct are recommended to be made at points 45° away from the vertical centerline of the duct. See Figure G-1 for an example.



NOTES:

1. Manifold tubing internal area shall be at least 4 times that of a wall tap.
2. Connecting tubing to pressure indicator shall be 6 *mm* (1/4 *in.*) or larger in ID.
3. Taps shall be within ± 13 *mm* (½ *in.*) in the longitudinal direction.

Figure G-1 Piezometer Ring Manifolding

This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ASHRAE Standard 51 but is included for information purposes.

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This Appendix is not a part of ANSI/AMCA Standard 210 or ANSI/ASHRAE Standard 51 but is included for information purposes.

APPENDIX I. HISTORY AND AUTHORITY OF ANSI/AMCA 210 - ANSI/ASHRAE 51, *LABORATORY METHODS OF TESTING FANS FOR AERODYNAMIC PERFORMANCE RATINGS*

This tenth edition of AMCA 210-ASHRAE 51 is the latest of a long series of Fan Test Codes that started with the first edition issued in 1923. Tradition has it that the first test code was developed as a result of problems encountered by the U. S. Navy with the performance ratings of fans being procured during World War 1. To resolve the issue of variations in testing methods which led to variations in performance ratings, a joint committee of the National Association of Fan Manufacturers (NAFM) and the American Society of Heating and Ventilating Engineers (ASHVE) was formed to develop a Standard Test Code for Fans.

The test code has been periodically reviewed and revised as problems arose and improvements to instrumentation and airflow measuring technology occurred. Over the years many people have made significant contributions to the development of this standard, some of whom are recognized in this history where appropriate. A brief outline of the significant changes that were made in each edition follows along with the complete Preface to the first edition.

Preface to the 1923 Edition of Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers

There are many instruments and methods existing today for use in testing of propeller fans, disc fans, and centrifugal fans and blowers. The different types of instruments used and the various methods or "setups" used have resulted in a wide variation in the data obtained.

Due to this lack of uniformity the Performance Tables and Characteristic Curves of fans and blowers have lacked uniformity and have not been comparable on a uniform basis.

Recognizing not only the desirability, but the necessity, of a standard method of testing fans and blowers in order to give them a proper rating, a Joint Committee was selected by the American Society of Heating and Ventilating Engineers and the National Association of Fan Manufacturers to prepare a Standard Test Code.

The Joint Committee was assisted by a sub-committee, composed of the Research Engineers of the member companies of the National Association of Fan Manufacturers.

The Joint Committee also cooperated with Committee No. 10 of the Power Test Code Committee of the American Society of Mechanical Engineers to the end that the respective codes would coincide where they covered the same ground.

The Standard Test Code for rating fans and blowers is offered to you with the belief that its use will be of marked benefit to both the manufacturer and user of fan and blower apparatus.

Joint Committee:

H. W. Page	B. F. Sturtevent Company
Prof. F. Paul Anderson (Secretary)	Director-Research Laboratory, American Society of Heating and Ventilating Engineers; Dean, University of Kentucky, College of Engineering
F. R. Still	American Blower Co.
C. A. Booth	Buffalo Forge Co.
E. M. Bassler	Bayler Manufacturing Co.
A. K. Owens	Tenny & Ohmes

Sub-Committee:

W. A. Rowe	American Blower Co.
J. C. Wolf	Bayley Mfg. Company
A. A. Criqui	Buffalo Forge Company
L. O. Monroe	Clarage Fan Company
E. D. Green	Garden City Fan Company
C. S. Messler	Green Fuel Economizer Co.
A. G. Sutcliffe	Ilg Electric Ventilating Co.
C. W. Rodgers	New York Blower Co.
H. F. Hagen	B. F. Sturtevent Company

The Test Code was revised in 1932 with the addition of provisions for testing fans with inlet boxes. In 1938 the Test Code was revised on the basis of research made with the help of various colleges of engineering, consulting engineers and other organizations. Flow straighteners were added to the test ducts, and the allowance for skin friction was reduced. Nomenclature changes relating to fan types and usage were made in the 1949 edition, which was published as NAFM Bulletin No. 110.

In 1955 NAFM was combined with the Power Fan Manufacturers Association (PFMA) and the Industrial Unit Heater Association (IUHA) to form the Air Moving and Conditioning Association (AMCA). One of the major concerns of this new organization was the accuracy and practicality of the Pitot traverse method of testing, and a committee was formed to study various test methods and to develop a new test code. To aid in this study AMCA sponsored research by the Battelle Memorial Institute to compare the test results using the Pitot tube test methods and nozzle test methods. The result of this effort was a new revision of the test code which was published in 1960 by AMCA as the AMCA Standard Test Code for Air Moving Devices, Bulletin 210. This fifth edition of the test code represented a major step forward in standard methods for testing fans and provides the underlying basis for all subsequent editions. The nozzle method of measuring air flow was recognized in this edition and the chamber-nozzle "setup" was developed and incorporated in the test code. Provision was also included for the effects of compressibility on fan performance.

The engineering committee that produced the fifth edition was composed of:
Tom Walters - American Standard Industrial Division
Bob Parker - Ilg Electric Ventilating Company
D. D. Herrman - Hartzell Propeller Fan Company
Hoy Bohanon - Acme Engineering & Manufacturing Co.

AMCA revised the Standard in 1960 to show arrangements for testing multiple outlet units. These changes, while minor, resulted in the sixth edition.

The seventh edition was published by AMCA in 1967 incorporating a number of changes in general format. International Standards were used as the basic units of measurement, and the symbols used throughout the code were consolidated into one table. As a result of AMCA sponsored research into the effectiveness of the Flow Straightener at unusual flow conditions, the cell size and length were changed, and a new derivation of the Compressibility Factor was added as an appendix.

AMCA Standard 210 became widely accepted and was virtually the only standard used in the United States and Canada since 1960. It had been widely accepted by producers, customers, and general interest groups, but no national consensus had ever been recorded. The Air Movement and Control Association (AMCA) asked The American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) to form a joint committee to facilitate obtaining national consensus. The Joint Committee first met on February 10, 1971 and decided that AMCA Standard 210-67 would be the starting point for development of a national consensus Standard. The following excerpts from the Foreword to the eighth edition published as AMCA Standard 210-74/ASHRAE Standard 51-75 provides an overview of the detailed review and revision of AMCA Standard 210-67.

Excerpts from the Foreword to AMCA Standard 210-74-/ASHRAE Standard 51-75:

The provisions of AMCA Standard 210-67 were subjected to critical review. The progress of the TC 117 Committee of the International Organization for Standardization (ISO) on methods of testing industrial fans, was monitored. The literature, particularly that on flow and pressure measurement, was searched. The Joint Committee recommended experimental work which was conducted in the AMCA Laboratory. All of this influenced the final content of this standard.

Some of the more significant differences between this standard and its predecessor, AMCA Standard 210-67, are:

(1) The style was changed to reflect American National Standards Institute (ANSI) recommendations regarding page format, abbreviations, symbols and subscripts, and general arrangement.

(2) The content of the standard is limited to matters directly related to testing. Other information, including the application of the fan laws for rating purposes, is contained in the appendices.

(3) The scope has been broadened by eliminating the upper limit on compression ratio. The scope has been narrowed by limiting the test gas to air.

(4) The units of measurement for gas properties are based on mass rather than weight. Water gauge is based on 68°F and includes a gas column balancing effect.

(5) The definitions have been expanded to include total temperature, head and compressibility coefficient.

(6) Test setups have been given new numerical designations.

(7) Performance specifications, as well as equipment specifications, are given for instruments and methods of measurements.

(8) A log-linear Pitot traverse method has been substituted for the equal area method.

(9) Data for nozzles have been expanded.

(10) Specifications for chamber size and chamber settling means have been changed.

(11) Calculation methods have been revised with respect to duct friction, straightener loss, compressibility coefficient, and conversion formulae. Calculation methods are presented in a manner to facilitate either manual or automatic data processing.

(12) Both the International System of Units (SI) and other metric units are treated in an appendix.

(13) The fan total efficiency equation for compressible flow is derived in an appendix. This is done in terms which eliminate the need for an iteration procedure.

(14) New compressible-flow fan laws are derived in an appendix.

(15) An error analysis method is derived in an appendix.

While each of the above changes is significant, the basic procedures of AMCA Standard 210-67 have been retained.

The Joint ASHRAE-AMCA Project Committee was composed of the following members:

Robert Jorgensen, Chairman	ASHRAE-AMCA
Kenneth W. Burkhardt, Secretary	AMCA
Nestor Brown, Jr.	ASHRAE
Hoy R. Bohanon	ASHRAE-AMCA
Harold F. Farquhar	ASHRAE-AMCA
Linn Helander	CONSULTANT
Donald D. Herman	AMCA
John G. Muirheid	ASHRAE
Allen C. Potter	ASHRAE
Wendell C. Zeluff	AMCA

E. A. Cruse, ASHRAE Standards Committee Liaison

In 1977 AMCA Standard 210-74/ASHRAE Standard 51-75 was granted American National Standards Institute status becoming ANSI/AMCA 210/ASHRAE 51-75.

The standard was reviewed by another Joint Committee starting in 1982 and was reaffirmed with minor revisions. Installation types corresponding with those in the British Standard BS848: Part 1:1980 and the draft ISO Standard (5801) were included, along with slight changes to the Inlet Bell and Settling Means. The appendix was changed to reflect the more acceptable term "uncertainty" as opposed to "error" and portions of the derivation were modified.

The ninth edition of the standard was published as ANSI/AMCA Standard 210-85 - ANSI/ASHRAE Standard 51-1985.

The Joint ASHRAE-AMCA Project Committee was composed of the following members:

Robert Jorgensen, Chairman	ASHRAE-AMCA
Gordon V.R. Holness, Vice Chairman	ASHRAE
Hoy R. Bohanon	ASHRAE-AMCA
A. Michael Emyanitoff	ASHRAE
Daniel Fragnito	ASHRAE
James W. Schwier	ASHRAE-AMCA
Thomas A. Hirsbrunner	ASHRAE-AMCA
Gerald P. Jolette	ASHRAE-AMCA

Since AMCA 210-67 was published, this standard has grown in recognition not only in the United States and Canada but also internationally. The provisions of this standard were included almost in its entirely in the International Standard, ISO 5801. The Foreword to this the tenth edition outlines the minor changes made to this edition. In the coming years the standard will again be reviewed and revised to reflect improvement to the technology of flow and pressure measurement, but it is always well to understand the foundation on which this standard was built and the people who contributed to its construction.

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Interpretation IC-51-1999-1 of
ANSI/ASHRAE Standard 51-1999/AMCA Standard 210-1999
Laboratory Methods for Testing Fans for Aerodynamic Performance Rating

February 18, 2000

Reference: This request refers to ANSI/ASHRAE 51-1999/AMCA 210-1999, Sub-clause 6.3.3

Background: The second paragraph of 6.3.3 reads:

"When a measuring plane is located downstream of the settling means, the settling means is provided to ensure a substantially uniform flow ahead of the measuring plane. In this case, the maximum velocity at a distance 0.1M downstream of the screen shall not exceed the average velocity by more than 25% unless the maximum velocity is less than 400 feet per minute."

Interpretation No. 1: Requestor's letter opines that this check is required at the design airflow (400 fpm x cross sectional area of the chamber) rating of the chamber.

Question No. 1: Is Interpretation No. 1 correct?

Answer: No.

Comment: The requirement must be met at all airflows used during the test.

Interpretation No. 2: Requestor's letter opines that this check is not required at intermediate air quantities if the check passes at the design airflow rating of the chamber.

Question No. 2: Is Interpretation No 2 correct?

Answer: No.

Comment: The check is required at all airflows used during the test.

Question No. 3: If the answer to Question No. 2 is NO and the intermediate air quality check is required, must more than one combination of nozzles be checked?

Answer: Yes.

Comment: Any combination of nozzles used during the test must be checked.

Interpretation No. 3: Requestor's letter opines that 30 points spaced in accordance with the log Tchebycheff rule for rectangular ducts (1993 ASHRAE Handbook, Figure 6, page 13.15) is adequate for this check.

Question No. 4: Is Interpretation No. 3 correct?

Answer: Yes.

Interpretation No. 4: Requestor's letter opines that an Electronic Air Velocity Indicator with an accuracy of ±3% (of reading) or ±10 fpm, whichever is greater, meets the requirements of this standard.

Question No. 5: Is Interpretation No. 4 correct?

Answer: Provided calibration records are in order.

Interpretation No. 5: Requestor's letter states that the local velocity at each point in the measurement fluctuates and opines that the velocity to be recorded is the average velocity, not the maximum velocity.

Question No. 6: Is Interpretation No. 5 correct?

Answer: Yes.

Comment: All local velocities have to be averaged in time as described in 6.2.1.2 of the standard.

Interpretation No. 6: Requestor's letter opines that the check is done with the control fan only (without a test fan).

Question No. 7: Is Interpretation No. 6 correct?

Answer: No.

Comment: The requirements must be met with the test fan operating. The control fan may or may not be operating.

Interpretation No. 7: Requestor's letter opines that the average velocity in 7.3.3 is that calculated by *V (fpm) = Q (cfm) / A (cross sectional area of chamber,* ft^2*)*

Question No. 8: Is Interpretation No. 7 correct?

Answer: Yes.

General Comment: The standard is written to cover the requirements for a test. Since a test must cover a range of airflow rates, any requirement predicated on airflow rate, such as average velocity or maximum velocity, must be met for the full range of airflow rates. For instance, if there are eight determinations made during a test, the requirement must be satisfied for each determination. Any test utilizing equipment based on Figures 9, 10, 11 or 15 must meet the requirements of the paragraph cited in this interpretation at any and all airflow rates being measured. There is no single design airflow rate for the apparatus according to this standard. The maximum velocity referred to in sub-clause 7.3.3 is the maximum of the various readings across the face of the settling means during a determination.

Interpretation IC 51-1999-2 - September 12, 2000
ANSI/ASHRAE STANDARD 51-1999 and ANSI/AMCA Standard 210-99
Laboratory Methods of Testing Fans for Aerodynamic

Reference: This request refers to Interpretation IC 51-1999-01 of ANSI/ASHRAE 51-1999 and ANSI/AMCA 210-99 dated February 18, 2000.

Background: Requestor cites the second paragraph of subclause 6.3.3,

> "When a measuring plane is located downstream of the settling means, the settling means is provided to insure a substantially uniform flow ahead of the measuring plane. In this case, the maximum velocity at a distance 0.1M downstream of the screen shall not exceed the average velocity by more than 25% unless the maximum velocity is less than 400 feet per minute."

and the following interpretations of this paragraph, provided in IC 51-1999-01:

> "(a) The requirement must be met at all airflows during a test."
>
> "(b) The check is required at all airflows used during the test."
>
> "(c) Any combination of nozzles used during the test must be checked."
>
> "(d) The requirements must be met with the test fan operating. The control fan may or may not be operating."
>
> "(e) The standard is written to cover the requirements for a test. Since a test must cover a range of airflow rates, any requirement predicated on airflow rate, such as average velocity or maximum velocity, must be met for the full range of airflow rates. For instance, if there are eight determinations made during a test, the requirement must be satisfied for each determination. Any test utilizing equipment based on Figures 9, 10, 11 or 15 must meet the requirements for the paragraph cited in this interpretation at any and all airflow rates being measured. There is no single design airflow rate for the apparatus according to this standard. The maximum velocity referred to in subclause 6.3.3 is the maximum of the various readings across the face of the settling means during a determination."

Question No. 1: If the above requirement is to be met at all airflow determinations, is it required to conduct a qualification test for **every fan test** to verify compliance with 6.3.3?

Answer: Yes, except when a qualification procedure has been agreed to as noted in the General Comment, below.

Question No. 2: If the above requirement is met by conducting tests at all airflows to verify compliance with 6.3.3, is the qualification data to be reported with every fan test report?

Answer: Yes, except when a qualification procedure has been agreed to as noted in the General Comment, below.

Question No. 3: Requestor opines that a laboratory pre-qualification test using a test fan having sufficient airflow capacity could be used as a basis for all other fans to be tested, thereby eliminating the check-test requirement for every fan at all airflow determinations. This way the total time to test fans would be within reasonable time periods. Are one-time pre-qualification tests acceptable to meet the requirements of subclause 6.3.3?

Answer: No, except when a qualification procedure has been agreed to as noted in the General Comment, below.

General Comment:

There is no provision in the Standard for general qualification of a test facility to meet the requirements of subclause 6.3.3.

The parties to a test may agree on a procedure to qualify the test facility over a range of airflows, nozzle combinations, outlet velocities of the test fan, supply fan and nozzles that meet the requirements of the standard. The parties to a test may further agree that any test performed within the qualification range of the test facility would be deemed to have met the requirements of the Standard and would not require additional checks.

Addendum a-2001
to
ASHRAE 51-1999 / AMCA 210-99

Approved by American National Standards Institute 21 August 2001

In revising the 1985 edition, one revision produced Note 4 of Figure 4A. The intended result of the revision was to specify a dimensional measurement location that would be suitable for checking by a state-of-the-art measurement device. The unintended effect of this revision was the tightening of the tolerance on that dimension beyond that which is possible to achieve by existing and customary commercial fabrication processes. This went unnoticed during the review process. To rectify this oversight, it is necessary to revert to the requirement as given in the 1985 edition:

"4. The nozzle throat shall be measured (to an accuracy of $0.001D$) at the minor axis of the ellipse and the nozzle exit. At each place, four diameters — approximately 45° apart must be within $\pm 0.002D$ of the mean. At the entrance to the throat the mean may be $0.002D$ greater, but no less than, the mean at the nozzle exit."

ERRATA SHEET FOR
ANSI/ASHRAE 51-1999 (ANSI/AMCA 210-1999)
Laboratory Methods of Testing Fans for Aerodynamic Performance Rating

February 15, 2006

The corrections listed in this errata sheet apply to all printings of ANSI/ASHRAE Standard 51-1999 (ANSI/AMCA 210-99). The shaded items have been added since the previously published errata sheet dated January 10, 2006 was distributed.

Page	Erratum
56	**Appendix F. Iterative Procedure:** In the first paragraph, first sentence, change the section referenced from "9.3.2.7" to "8.3.2.7".
56	**Appendix F. Iterative Procedure:** In Equation F-1 (first column of page 56) the value of Y (expansion factor) should be calculated per Section 8.3.2.3, not Section 9.3.2 as indicated. Change "Y = 0.998 (calculate per Section 9.3.2)" to read "Y = 0.998 (calculate per Section 8.3.2.3)".
56	**Appendix F. Iterative Procedure:** In Equation F-2 I-P (second column of page 56) change the number in the square root in the denominator from "195595" to "189595". The calculated value of Ce_1 (= 0.98322) is correct.

AIR MOVEMENT AND CONTROL
ASSOCIATION INTERNATIONAL, INC.

30 West University Drive Arlington Heights, Illinois, 60004-1893 USA
Phone: 847/394-0150 · Fax: 847/253-0088 · www.amca.org

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